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PRELIMINARY DESIGN OF A MECHANICAL-DRAFT COUNTERFLOW COOLING TOWER

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ARO, Inc.

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NOTICES

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FOREWORD

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This technical report has been reviewed and is approved.

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ABSTRACT

The need for a method whereby the preliminary design of a cooling tower can more easily be specified has existed for many years. The numerical techniques commonly used to solve the Merkel cooling tower equation are not adaptable to a direct general solution. This report presents a direct solution to the Merkel equation which, when combined with certain detailed experimental data by others, yields a relatively quick and straightforward method of establishing the preliminary design of a mechanical-draft counterflow cooling tower. Special attention is devoted to cooling tower applications relative to high altitude engine test facilities.

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NOMENCLATURE

A, b, d, e, n	Constants
a	Water surface area per unit tower volume, ft^2/ft^3
C	Constant of integration
c	Specific heat, Btu/(lbm)(°F)
D	Discriminant of the quadratic expressing the enthalpy potential
G	Rate of airflow, lbm/hr
h	Enthalpy, Btu/lbm

K	Overall unit conductance, bulk water to bulk air, lbm air/ft^2 (water area) hr, evaluated at bulk water temperature
K_G	Unit conductance, sensible heat transfer between interface and main air stream, Btu/hr-ft^2-°F
K_M	Unit conductance, mass transfer, interface to main air stream, lbm air/hr-ft^2
K_W	Unit conductance, heat transfer, bulk water to interface, Btu/hr-ft^2-°F
L	Rate of water flow, lbm/hr
m	Mass, lbm
N	Number of decks in tower fill
q	Rate of heat transfer, Btu/hr
r	Latent heat of vaporization for water, Btu/lbm
SH	Specific humidity, lbm water vapor/lbm dry air
t	Temperature, °F
V	Effective cooling tower volume, ft^3
Z	Effective cooling tower height, ft
Δ	Difference

SUBSCRIPTS

a	Air
i	Initial
L	Latent
m	Mixture
p	Constant pressure
s	Sensible
t	Total
w	Water
wb	Wet bulb
1	Condition No. 1
2	Condition No. 2

SUPERSCRIPTS

$'$	Defined in Figs. 2 and 3
$''$	Defined in Figs. 2 and 3

SECTION I
COOLING TOWER PRINCIPLES

A cooling tower is a structure whose function is to cool water through a certain temperature range by exposing the water to ambient air in a combined evaporation and convection heat transfer process. The structure is so designed that the warm water falls from the top to the bottom in thin curtains or droplets while the air enters from the bottom and flows to the top (vertically) for counterflow cooling or from the side (horizontal flow) for crossflow cooling (Fig. 1, Appendix).

Properly designed cooling towers ensure that a relatively large water surface is exposed to the air. This surface is produced by water dropping or splashing through a latticework called "fill" or "packing" usually constructed of decay-resistant wood.

Cooling towers are generally classified in two main groups: natural-draft and mechanical-draft.

Natural-draft towers depend on ambient conditions to produce a cooling airflow and are further subdivided: (1) the atmospheric tower in which prevailing winds provide all the ventilation, (2) the straight-wall chimney type which is rapidly becoming obsolete, and (3) the hyperbolic-wall tower. Both the chimney and hyperbolic type towers have sufficiently large heights such that airflow is established because of the difference in the density of the heated air inside the tower and the ambient air outside.

Mechanical-draft towers create their own air movement by means of a motor-driven fan. There are two primary classifications for these types of towers: (1) forced and (2) induced. The principal difference in the forced and induced towers lies in the location of the fan and auxiliaries. The forced-draft tower has the fan mounted on the lower side of the tower, where the air is blown in, allowing a more firm fan equipment support, thus reducing vibration. The induced-draft tower has the fan mounted on top of the tower to draw the air out. Since some of the fan velocity pressure is converted to static pressure in the forced-draft system, the exit velocity of the air is less than that of the induced type, thus allowing a greater degree of recirculation of the hot humid exhaust air. Exhaust vapors usually leave the forced-draft tower at such a low velocity that a cross wind causes them to be drawn back in the fan suction, thus artificially raising the wet-bulb temperature of the entering air which results in a higher cold water temperature. It is primarily for this reason that the forced-draft tower is rapidly losing favor in

industrial applications. In induced-draft towers, the exhaust air leaves at a much higher velocity and usually at a higher elevation (through the use of exhaust stacks) and the tendency to recirculate is reduced. Therefore, induced-draft towers are always used when it is desirable to maintain the lowest possible cold water temperature.

Mechanical-draft towers in general and induced mechanical-draft towers in particular have many advantages over other types of cooling equipment: (1) the area required is significantly less than that for an atmospheric tower, (2) cold water temperatures are more stable and predictable, (3) independence of wind which in turn allows (4) freedom of choice in location, and (5) minimum of drift nuisance when compared to the atmospheric tower which in turn leads to (6) smaller make-up water requirements. Disadvantages include higher first cost (except the hyperbolic type) and higher operating and maintenance cost.

In cases where a stable water exit temperature is needed, natural draft towers are almost never used because changes in wind velocity and direction cause the cold water temperature to fluctuate. Under certain extreme ambient conditions, this becomes especially important in high altitude engine test facilities where fluctuations in cold water temperature can lead to undesirable changes in simulated altitudes. A change in the cold water temperature of the exhaust gas surface cooler serviced by the cooling tower in such installations leads to a change in temperature of the exhaust gases being cooled. This, in turn, will change the performance of the exhaust compressor system which leads directly to a change in exhaust pressure (altitude).

In applications where large heat loads must be dissipated, natural draft towers (except hyperbolic) are almost never found. Again in relation to high altitude test facilities where large heat loads are commonly encountered, the required physical size and associated cost of a natural-draft tower would be prohibitive.

The hyperbolic tower, whose European popularity has been high for many years, is now increasing in favor in the United States. Enormous in size and heat load capability, these towers have the advantage of no moving parts and hence low operating and maintenance cost. The high initial installation cost is usually justified in industrial applications such as electric power plants where continuous operation is a necessity and shutdown due to mechanical failures must be avoided.

Hyperbolic towers and their associated higher initial cost cannot be justified at high altitude engine test facilities because continuous operation is unnecessary and occasional failures connected with mechanical-draft towers can be tolerated.

SECTION II
COOLING TOWER THEORY

2.1 THE MERKEL EQUATION

The generally accepted concept of cooling tower performance was developed by Merkel in 1925. The Merkel equation results from an analysis which combines the sensible and latent heat transfer into an overall process based on enthalpy potential as the driving force. Figure 2 shows the process schematically where each particle of the bulk water in the tower is assumed to be surrounded by an interface to which heat is transferred from the water.

The process will reach equilibrium when $t_a = t_w$ and the air becomes saturated with moisture at that temperature. Under adiabatic conditions, equilibrium is reached at the temperature of adiabatic saturation, or at the thermodynamic wet-bulb temperature of the air.

The total heat transfer from the water is the sum of the sensible and latent portions:

$$dq_t = dq_s + dq_L = K_G\,(a\,dV)\,(t' - t_a) + rK_M\,(a\,dV)\,(SH'' - SH_a) = G\,dh_a$$

Merkel utilized the Lewis relationship,

$$\frac{K_G}{K_M c_{pm}} \cong 1$$

in the development of the final equation:

$$\begin{aligned} dq_t = dq_s + dq_L &= K_M c_{pm}\,(a\,dV)\,(t' - t_a) \\ &\quad + rK_M\,(a\,dV)\,(SH'' - SH_a) = G\,dh_a \\ &= K_M\,(a\,dV)\left[c_{pm}\,t' + rSH'') - (c_{pm}\,t_a + rSH_a)\right] = G\,dh_a \end{aligned}$$

but, since enthalpy for an ideal gas mixture of air and water vapor is defined as

$$h = c_p\,t + rSH$$

then

$$dq_t = dq_s + dq_L = K_M\,(a\,dV)\,(h'' - h_a) = G\,dh_a \qquad (1)$$

The enthalpy potential in Eq. (1) is from the interface to the airstream which is indeterminate. This difficulty is overcome by ignoring the resistance of the film and considering an overall coefficient, K, which relates the driving force for the process to the enthalpy, h′, at the bulk water temperature, t_w.

The total heat transfer must also equal the total heat removed from the water. By neglecting the water loss by evaporation and assuming c_{pw} equals unity, the equation now becomes

$$L\ dt_w = G\ dh_a = K\ (a\ dV)\ (h' - h_a)$$

which upon integration yields

$$\frac{KaV}{L} = \int_{t_{w_2}}^{t_{w_1}} \frac{dt_w}{(h' - h_a)} \tag{2}$$

$$\frac{KaV}{G} = \int_{h_{a_2}}^{h_{a_1}} \frac{dh_a}{(h' - h_a)} \tag{3}$$

Figure 3 illustrates the cooling process in a counterflow cooling tower. As the water is cooled from temperatures corresponding to A to B, respectively (cooling range), the film enthalpy follows the saturation curve from C to D. The entering air at t_{wb} has as enthalpy corresponding to B′ which has the same value of enthalpy as point B. The difference between t_{w_2} and t_{wb} is called the approach temperature. Since,

$$L\ dt_w = G\ dh_a \tag{4}$$

the slope of line BA, dh_a/dt_w, is equal to L/G.

2.2 DIRECT SOLUTION OF THE MERKEL EQUATION FOR COUNTERFLOW COOLING

In the past, investigators have resorted to time-consuming numerical integration techniques in solving Eq. (2) for the design of a counterflow cooling tower. In addition, this indirect method of solution is very cumbersome and sometimes impossible to apply to the analysis of an existing tower to solve for certain critical parameters. Therefore, some direct method of solution is needed.

Close inspection of Eq. (2) and Fig. 3 will reveal that, if the enthalpy potential, $h' - h_a$, could be expressed as a function of water

temperature, t_w, then Eq. (2) could be integrated to obtain a direct solution. The difficulty of the integration process and the application of the resulting solution would, of course, depend on the mathematical complexity of the enthalpy potential.

Since, from Eq. (4),

$$\frac{dh_a}{dt_w} = \frac{L}{G}$$

integration yields

$$h_a = C + L/G\ t_w \tag{5}$$

At a water temperature of t_{w_2}, h_a must equal h_{ai}. Therefore, C must equal h_{ai} - $L/G\ t_{w_2}$ and the resulting equation for h_a is

$$h_a = (h_{ai} - L/G\ t_{w_2}) + L/G\ t_w$$

or

$$h_a = h_{ai} + L/G\ (t_w - t_{w_2}) \tag{6}$$

where h_{ai} is a function of ambient wet-bulb temperature, t_{wb}, only.

Obtaining a relationship for h′ as a function of water temperature is much more difficult and time consuming. The final relationship results from the method of "curve-fitting" which involves the assumption of general equation form and then fitting the arbitrary constants to form the particular solution.

After many assumptions, the resulting equation for h′ is a quadratic in t_w:

$$h' = \frac{{t_w}^2}{47.2} - 2.366\ t_w + 96.91 \tag{7}$$

The equation is within 1.5-percent accuracy in the cooling range of $75 \leq t_w \leq 120°F$. By combining Eqs. (6) and (7), the enthalpy potential can now be expressed in terms of t_w:

$$h' - h_a = \frac{{t_w}^2}{47.2} - (2.366 + L/G)\ t_w + (96.91 + L/G\ t_{w_2} - h_{ai})$$

Substitution into Eq. (2) gives

$$\frac{KaV}{L} = \int_{t_{w_2}}^{t_{w_1}} \frac{dt_w}{(h' - h_a)}$$

$$= \int_{t_{w_2}}^{t_{w_1}} \left[\frac{47.2\ dt_w}{t_w{}^2 - 47.2(2.366 + e)\ t_w + 47.2\ (96.91 - d)} \right] \tag{8}$$

where

$d = h_{ai} - L/G\ t_{w_2}$

$e = L/G$

Integration yields

$$\frac{KaV}{L} = \frac{47.2}{\sqrt{D}}\ \ell n \frac{\left| \dfrac{2t_{w_1} + b - \sqrt{D}}{2t_{w_1} + b + \sqrt{D}} \right|}{\left| \dfrac{2t_{w_2} + b - \sqrt{D}}{2t_{w_2} + b + \sqrt{D}} \right|} \tag{9}$$

when $D > 0$, or

$$\frac{KaV}{L} = \frac{47.2(2)}{\sqrt{-D}} \left[\tan^{-1} \frac{(2t_{w_1} + b)}{\sqrt{-D}} - \tan^{-1} \frac{(2t_{w_2} + b)}{\sqrt{-D}} \right] \tag{10}$$

when $D < 0$, or

$$\frac{KaV}{L} = -47.2(2) \left[\frac{1}{(2t_{w_1} + b)} - \frac{1}{(2t_{w_2} + b)} \right] \tag{11}$$

when $D = 0$, where,

$D = b^2 - 4c$

$b = -47.2\ (2.366 + e)$

$c = 47.2\ (96.91 - d)$

In order to obtain the correct solution from the three possibilities, reference is made to Fig. 3. Point A is always less than the corresponding point C for any operating condition because it is impossible for the wet-bulb temperature of the exhaust air to be equal that of the hot water. Point B is always less than point D because a zero approach temperature

can never be achieved in practice. Furthermore, for a good design, all the intermediate points on line BA will be less than those on DC. In relation to Eq. (8), this means that the discriminant, D, of the denominator must always be negative, and the proper solution is

$$\frac{KaV}{L} = \frac{47.2(2)}{\sqrt{-D}} \left[\tan^{-1} \frac{(2t_{w_1} + b)}{\sqrt{-D}} - \tan^{-1} \frac{(2t_{w_2} + b)}{\sqrt{-D}} \right]$$

The cumbersome numerical integration technique has thus been avoided, and a direct general solution is available. Figures 4 through 15 present the solution of Eq. 10 for some typical design conditions.

2.3 TOWER CHARACTERISTICS: REQUIRED AND AVAILABLE

When Eq. (10) is solved for a set of design conditions, the resulting value for KaV/L is referred to as the Number of Tower Characteristics (NTC)[1] required for the conditions selected. The required NTC represents a "degree of difficulty" for the conditions selected. When the same calculation is applied to a set of test data, the result is the available coefficient for the tower operating at those conditions under which the data were taken. Because of the idiosyncrasies in the behavior of the atmosphere in the vicinity of the cooling tower, the available coefficient at design conditions is extremely difficult to obtain. Methods have been proposed, however, which allow the "off-design" available coefficient to be corrected to obtain predicted cooling tower performance at design conditions.

A very detailed set of test data (Ref. 1) for ten different representative industrial fill geometries has shown that for any given geometry the available coefficient can be expressed in terms of the height of the filled section (effective height) and the water-to-air ratio, L/G:

$$KaV/L = 0.07 + AN\,(L/G)^{-n}$$

where A and n are constants and N is the number of decks in the fill. When information relative to the vertical deck spacing for the ten geometries are included and the two geometries with the highest and lowest

[1]Some authors prefer to call this the Number of Transfer Units (NTU). However, NTU is the value for KaV/G (named after Colburn) and NTC is KaV/L (named after Lichtenstein). Obviously, the numberical value differs only by the L/G ratio.

performance are omitted, the available coefficient for the eight remaining geometries can be expressed as

$$\frac{KaV}{L} = 0.07 + 0.0628\ Z \qquad (L/G = 1.0) \qquad (12)$$

$$\frac{KaV}{L} = 0.07 + 0.0565\ Z \qquad (L/G = 1.2) \qquad (13)$$

$$\frac{KaV}{L} = 0.07 + 0.0515\ Z \qquad (L/G = 1.4) \qquad (14)$$

within 17-percent maximum variation from the average. The maximum variation occurs, of course, for the two extreme geometries, and the variation for the remaining six is less. Figure 16 presents KaV/L versus height for the L/G ratios of 1.0, 1.2, and 1.4.

Equations (12), (13), and (14) along with Eq. (10) can be of particular value to the design engineer during initial preliminary investigation. Once the design conditions have been selected, Eq. (10) will predict the required coefficient. The substitution of the required coefficient for the available coefficient into the appropriate Eq. (12), (13), or (14) will yield an approximation of the necessary effective tower height.

SECTION III
APPLICATIONS OF THEORY TO THE PRELIMINARY DESIGN OF A MECHANICAL-DRAFT COUNTERFLOW COOLING TOWER

With the exception of the Merkel equation, the equations developed in Section II are valid only for counterflow cooling. The crossflow process, with its associated complexity of analysis, has thus far resisted a direct theoretical solution. In such cases, numerical or graphical techniques are employed. However, when the flow of air through a crossflow cooler approximates the counterflow path, the results presented herein can be used to obtain a rough preliminary estimate of crossflow cooling performance.

Normal design parameters for a cooling tower are: (1) wet-bulb temperature, (2) total heat load, (3) cooling range, and (4) approach. The equipment to be serviced by the tower and the ambient weather conditions will dictate the magnitude of each of the four parameters.

The selection of the design wet-bulb temperature is certainly critical and is almost always compromised by economic considerations.

For a constant approach, range, and L/G ratio, the higher the design wet-bulb temperature is, the lower the required tower characteristic, KaV/L, required to accomplish the cooling. This is primarily because for increased temperature the air has an increased capability for evaporative cooling. Even though the required tower characteristic will decrease with increasing wet-bulb temperature, the parameter sacrificed will be the cold water temperature which for the same approach will increase with increased design wet-bulb temperature. Therefore, from a performance standpoint a LOW design wet-bulb temperature is desirable because it produces a higher required tower characteristic and thus increases the number of hours per year that the tower will operate within the guarantee zone. A common practice is to select a wet-bulb temperature that will prevail or be exceeded 5 percent of the time during the four summer months. Information relative to design wet-bulb temperature for specific locations in the United States may be found in Ref. 2.

By neglecting the usually small heat losses to the ambient air, the total heat load is identically equal to the heat produced by the equipment being serviced by the cooling tower.

Because the specific heat of liquid water is approximately unity, the cooling range is related directly to the total heat load through the total water flow rate:

$$q_t = L\,\Delta t_w = L\,(\text{range})$$

or

$$\text{range} = q_t/L$$

The total water flow rate, L, must be selected sufficiently large to maintain the range within reasonable limits. Consideration of other components of the entire cooling system must also be made in determining L. Mechanical-draft cooling towers are most commonly designed for a range, Δt_w, of between 20 and 40°F. As the range is increased, the required tower characteristic is increased, other parameters remaining constant.

Cooling tower design is very sensitive to approach temperature. As the approach temperature is decreased, the required tower characteristic increases sharply. It is only when the particular application dictates that the lowest possible cold water temperature be maintained that a small approach temperature is selected. Common cooling tower designs limit the minimum approach temperature to approximately 5°F. In cooling tower application to high altitude engine test facilities, the stability of

the cold water temperature is more important, within reasonable limits, than the magnitude, and hence, a higher approach temperature of 10 to 15°F should be selected.

Figures 4 through 15 present the required tower characteristic in terms of the parameters previously discussed and the water-to-air ratio, L/G. It should be noted that the required NTC is independent of the absolute magnitudes of L and G and dependent only on the ratio L/G. In actual practice, however, both L and G are limited. Economic considerations involving capital charges and fan power costs limit the fan power and the resulting airflow rate to about 1800 lbm/hr-ft^2 of effective tower area. If the water flow is increased beyond around 3000 lbm/hr-ft^2, the water tends to cascade in streams so that the effective surface area is reduced. Furthermore, if the water flow drops to about 600 lbm/hr-ft^2, surface tension causes the water flow to channel. For preliminary design, it is suggested that a water loading of about 2160 lbm/hr-ft^2 in conjunction with an air loading of 1800 lbm/hr-ft^2 be used, resulting in an L/G ratio of 1.2. As the state-of-the-art progresses and refinements are made in tower fill material and geometry, higher loadings can be expected.

After the design parameters have been selected, the preliminary cooling tower design can be established as follows:

1. Equation (10) or the appropriate Fig. 4 through 15 yields the required tower characteristic.
2. With the required tower characteristic, enter Fig. 16 and obtain the required effective cooling tower height (height of the filled section).
3. The quotient resulting from the total water flow, L, and the water loading of 2160 lbm/hr-ft^2 gives the necessary effective plan area.
4. The plan area in conjunction with an air loading of 1800 lbm/hr-ft^2 results in the total airflow, G, which gives an indication of fan size and quantity.
5. The appropriate Fig. 17 through 19 yields the discharge air temperature.

It should be emphasized that, even though the design parameters establish a design point, the cooler will operate at any point along the constant NTC line predicted in step No. 1 above. As the warm water temperature is changed, the cooling tower will simply readjust to a different approach temperature and range at the same value of NTC.

Because the NTC is essentially constant for the same L/G ratio, the selection of a design point results automatically in the selection of a design line.

SECTION IV
CONCLUDING REMARKS

The information contained in this report should prove to be very useful to the design engineer in obtaining the preliminary design of a mechanical-draft counterflow cooling tower. By using this method, an estimate of the required physical size of a cooling tower can be obtained rather quickly, thus allowing important conclusions to be made concerning the feasibility of a cooling tower installation to the particular application.

In addition to being straightforward and much less time-consuming, the development of this design method, with the exception of Eqs. (12) through (14), is more accurate than the commonly used numerical technique. Even the 17-percent maximum variation contained in Eqs. (12) through (14) is usually within the range of engineering accuracy for preliminary calculations. When establishing a more detailed design, the design engineer could eliminate this approximation by selecting a specific fill geometry and then utilizing Ref. 1 to determine the exact fill performance.

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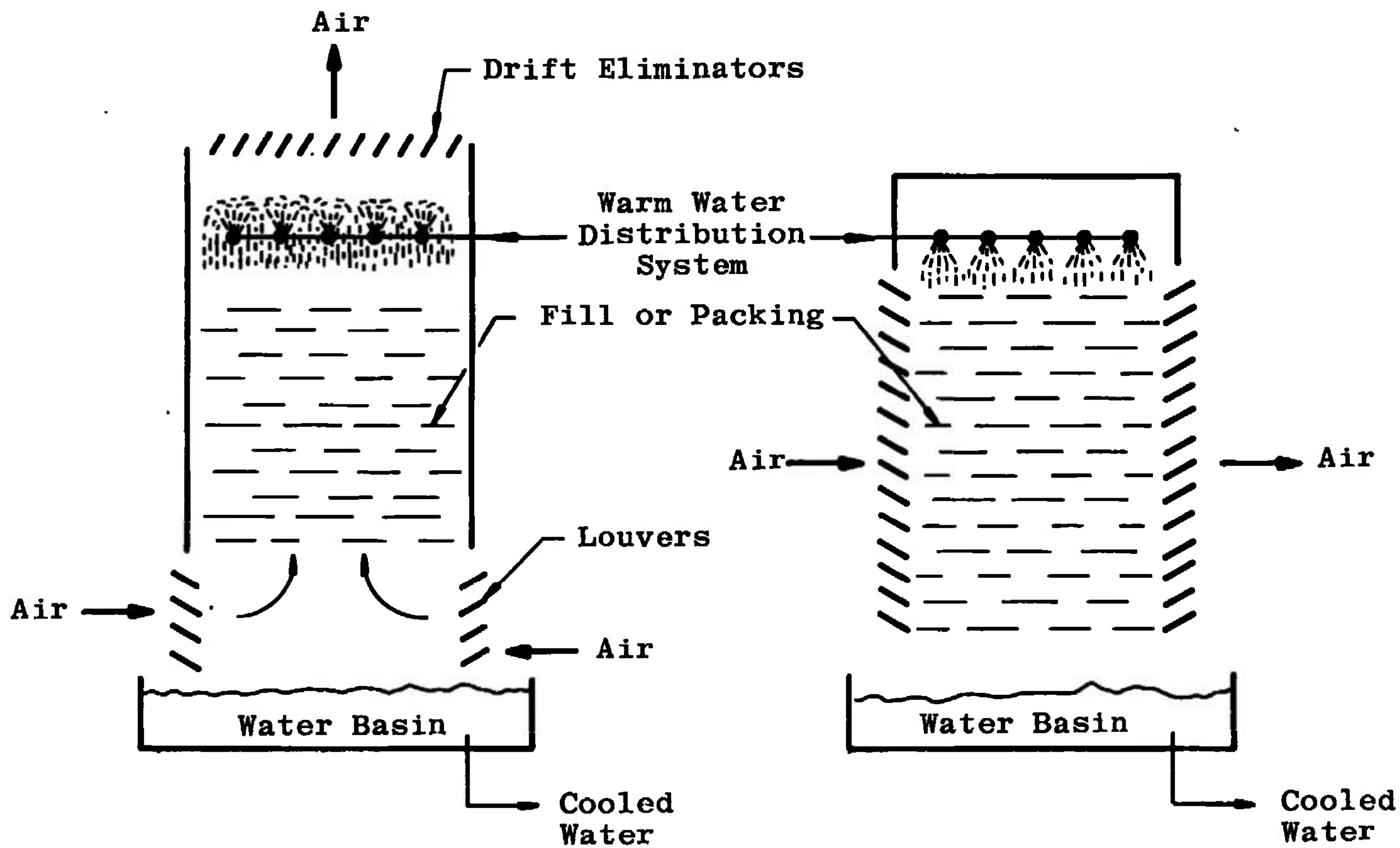
APPENDIX
ILLUSTRATIONS



a. Counterflow Cooling

b. Crossflow Cooling

Fig. 1 Cooling Towers: Basic Cooling Methods

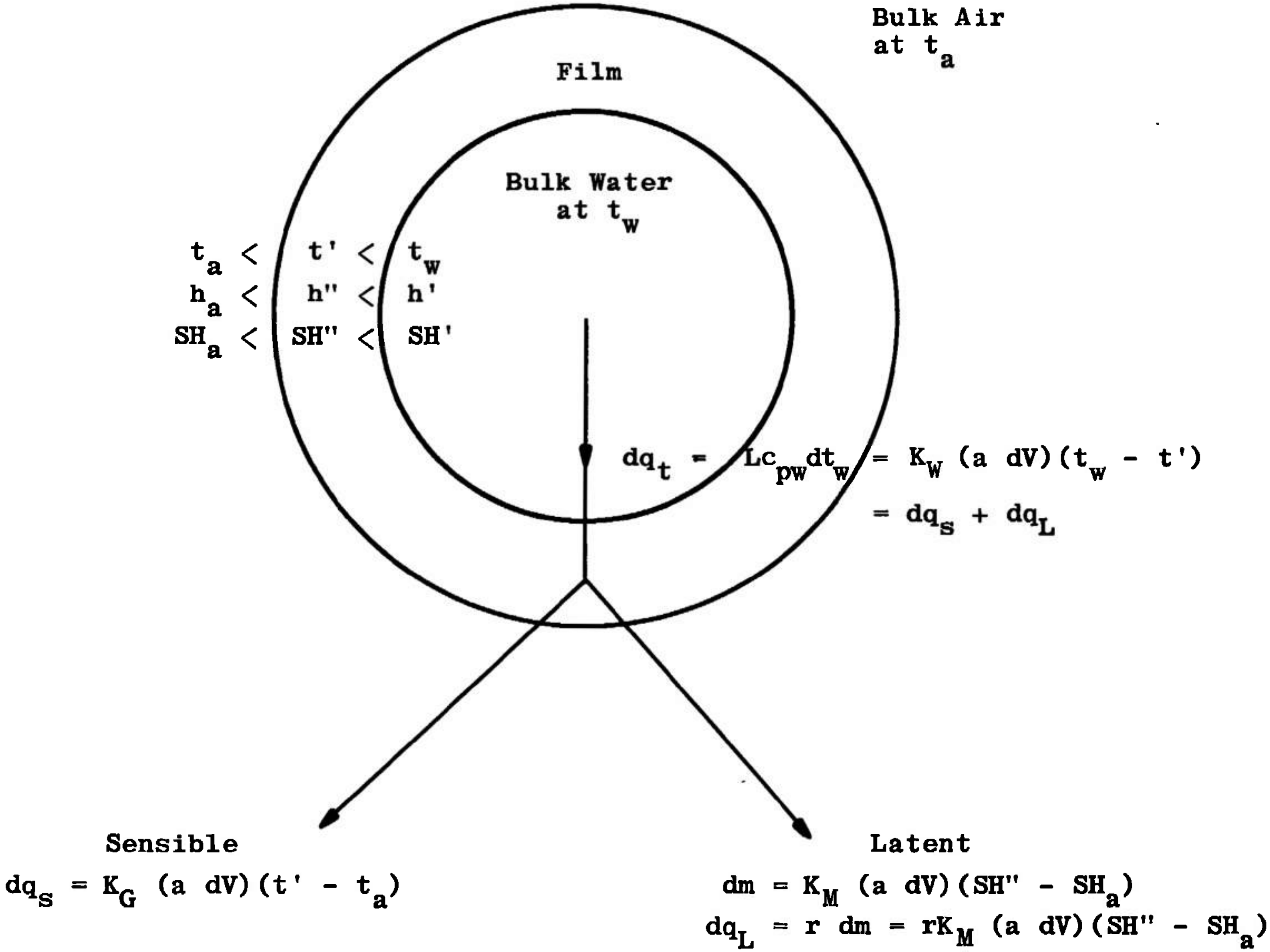


Fig. 2 Heat and Mass Transfer Relationships between Water, Film, and Air

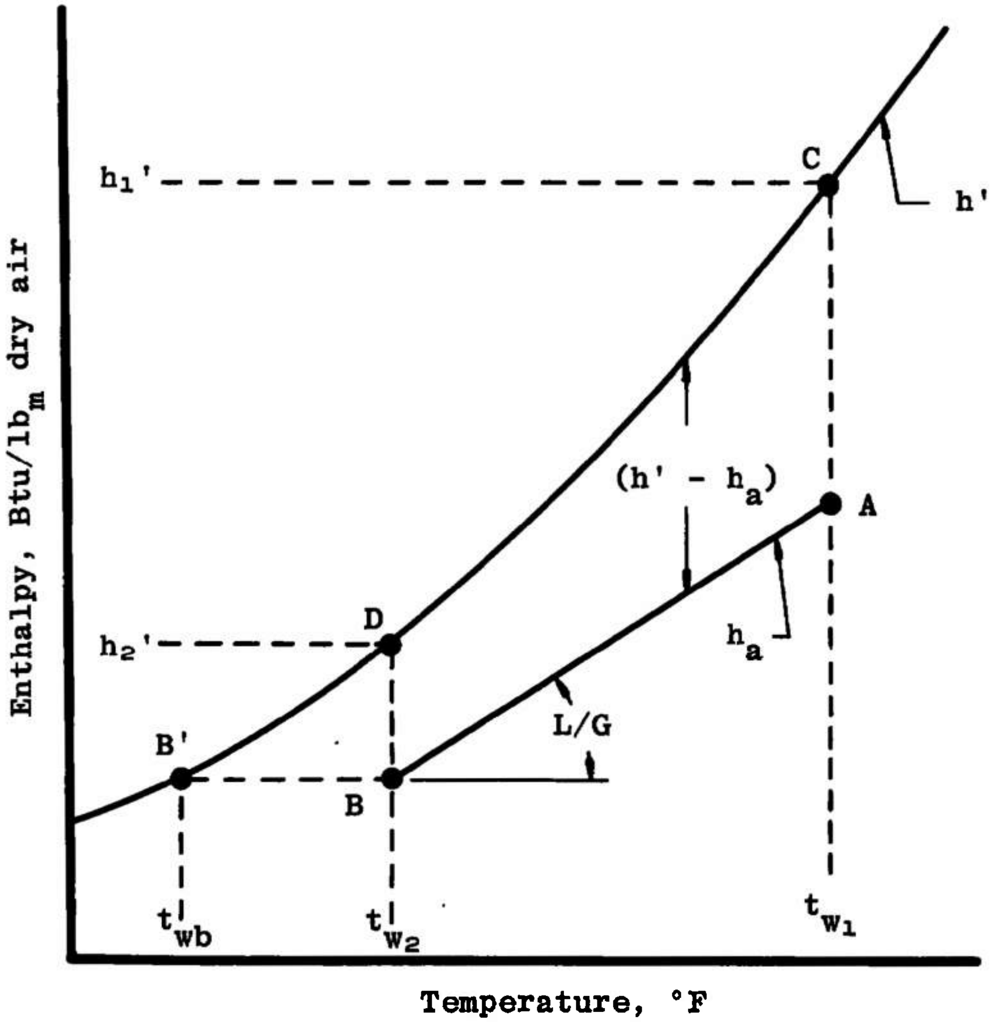


Fig. 3 Counterflow Cooling

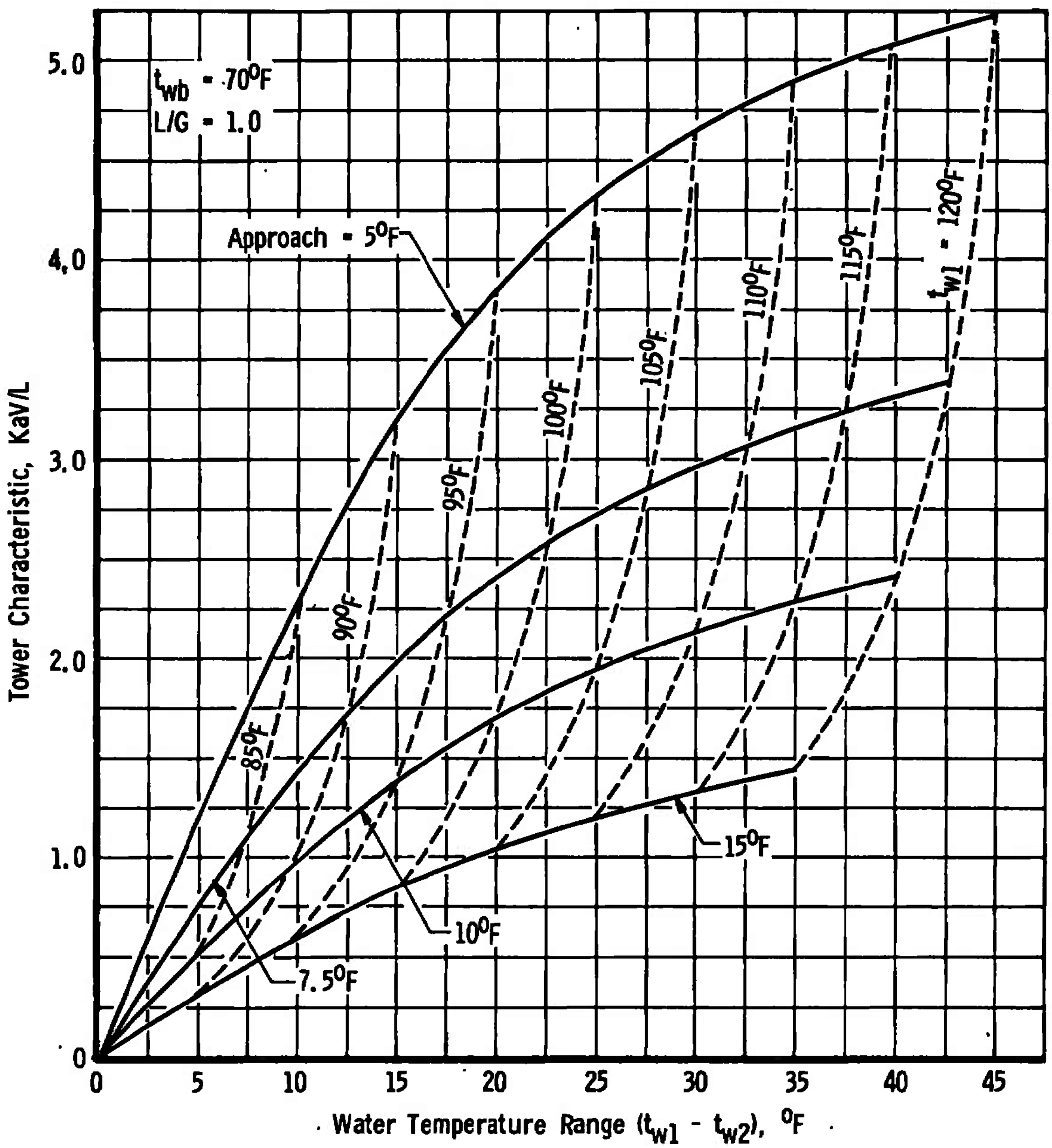


Fig. 4 Tower Characteristic versus Water Temperature Range for t_{wb} = 70°F and L/G = 1.0

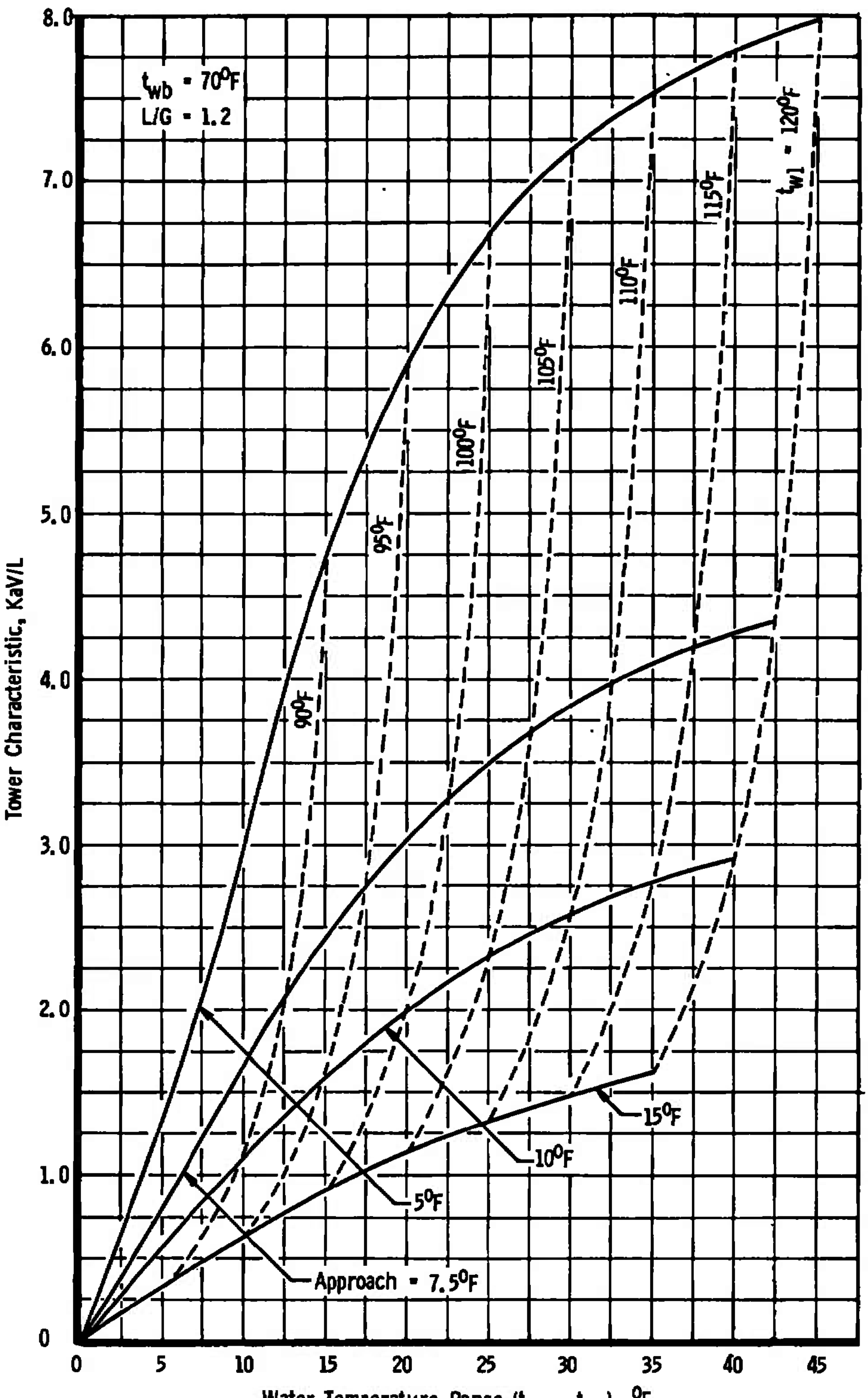


Fig. 5 Tower Characteristic versus Water Temperature Range for t_{wb} = 70°F and L/G = 1.2

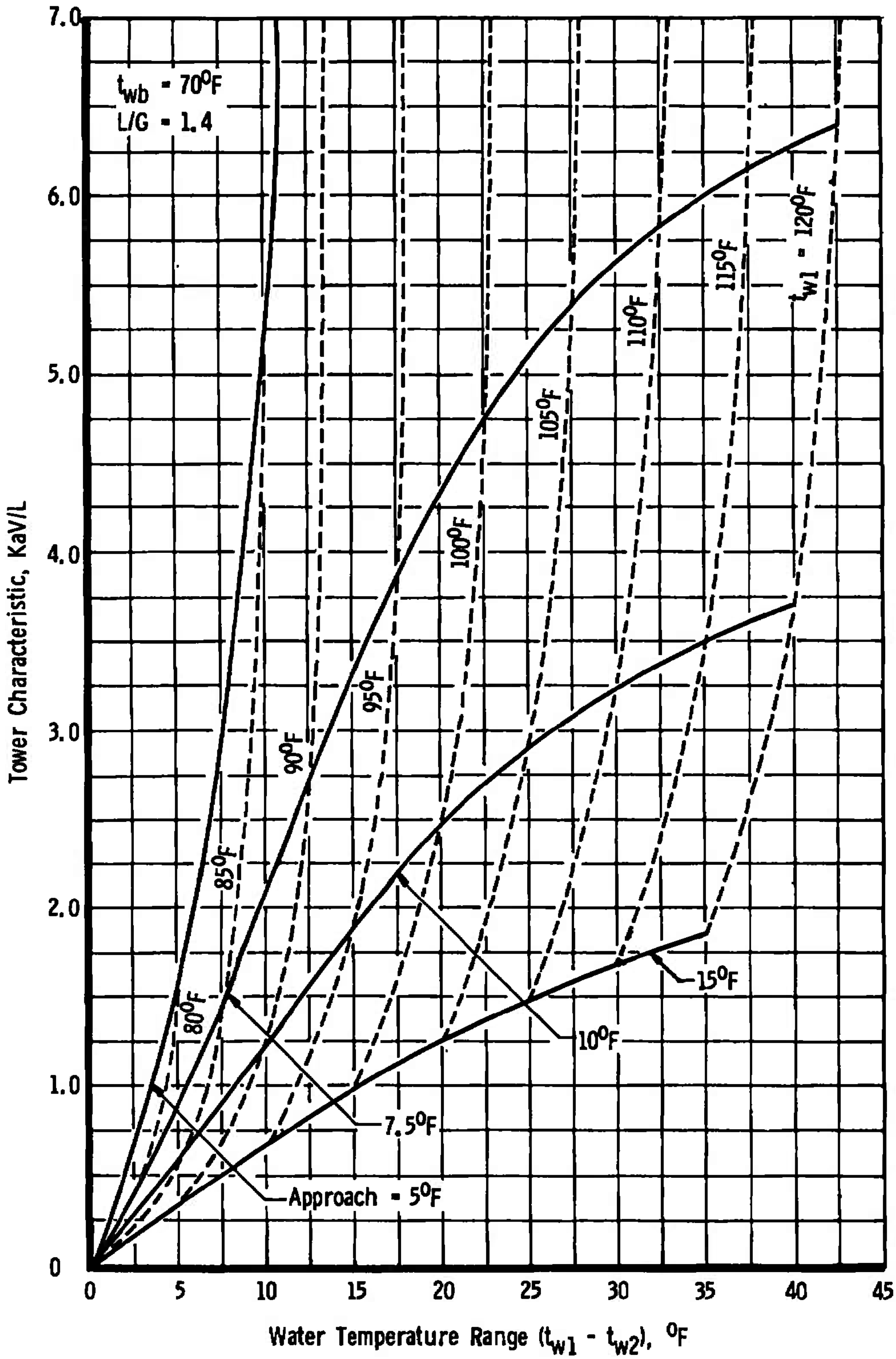


Fig. 6 Tower Characteristic versus Water Temperature Range for t_{wb} = 70°F and L/G = 1.4

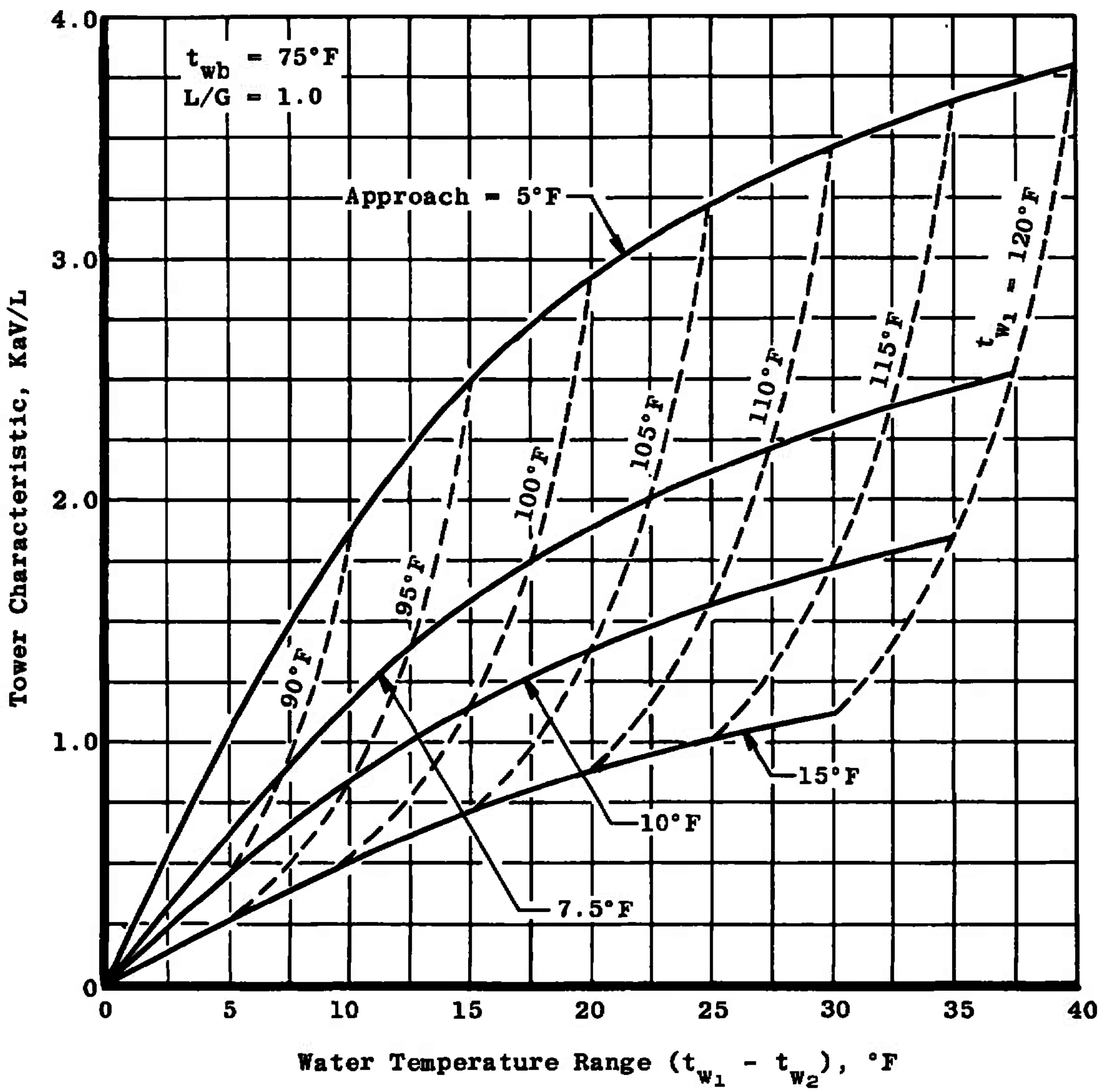


Fig. 7 Tower Characteristic versus Water Temperature Range for t_{wb} = 75°F and L/G = 1.0

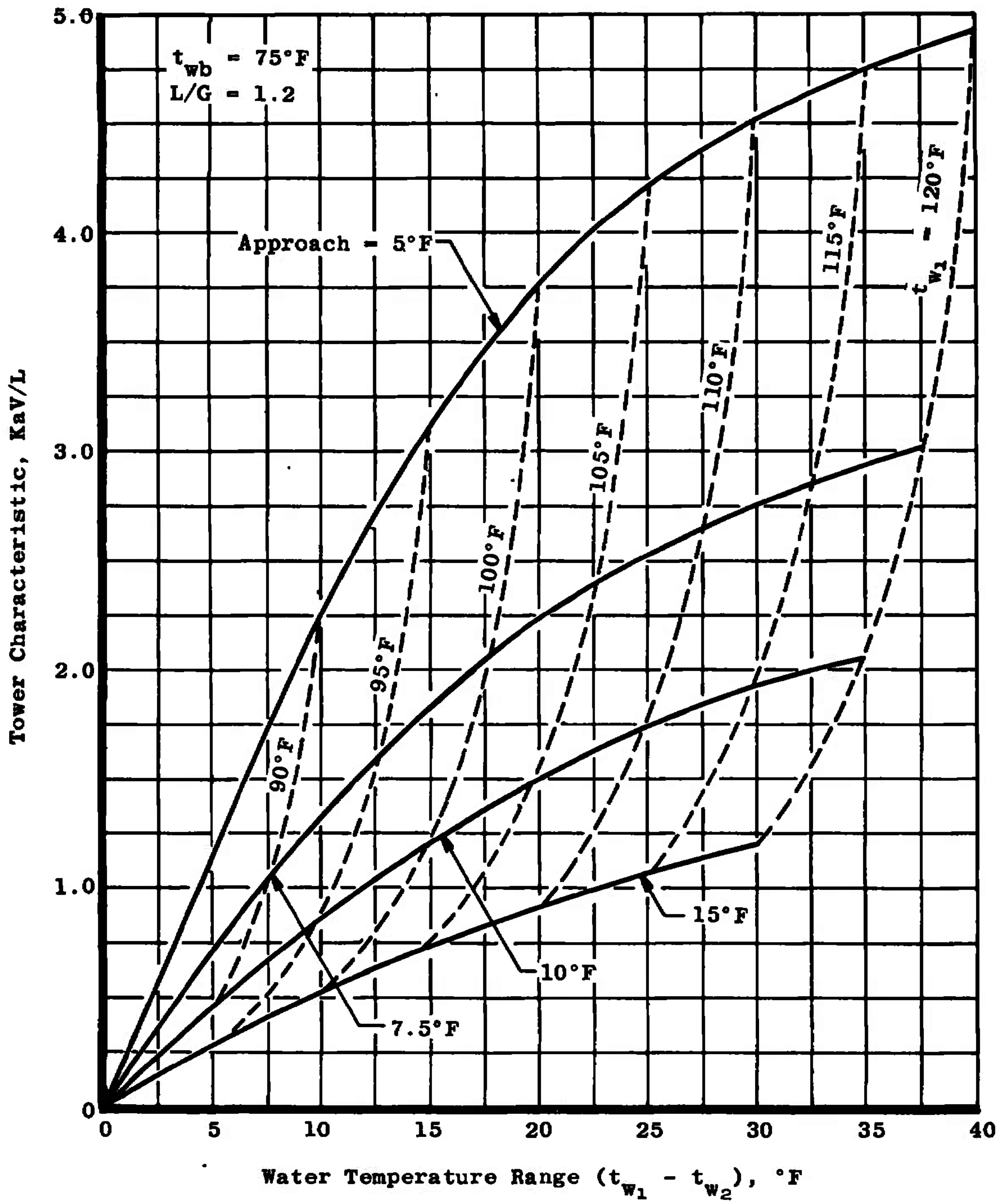


Fig. 8 Tower characteristic versus Water Temperature Range for t_{wb} = 75°F and L/G = 1.2

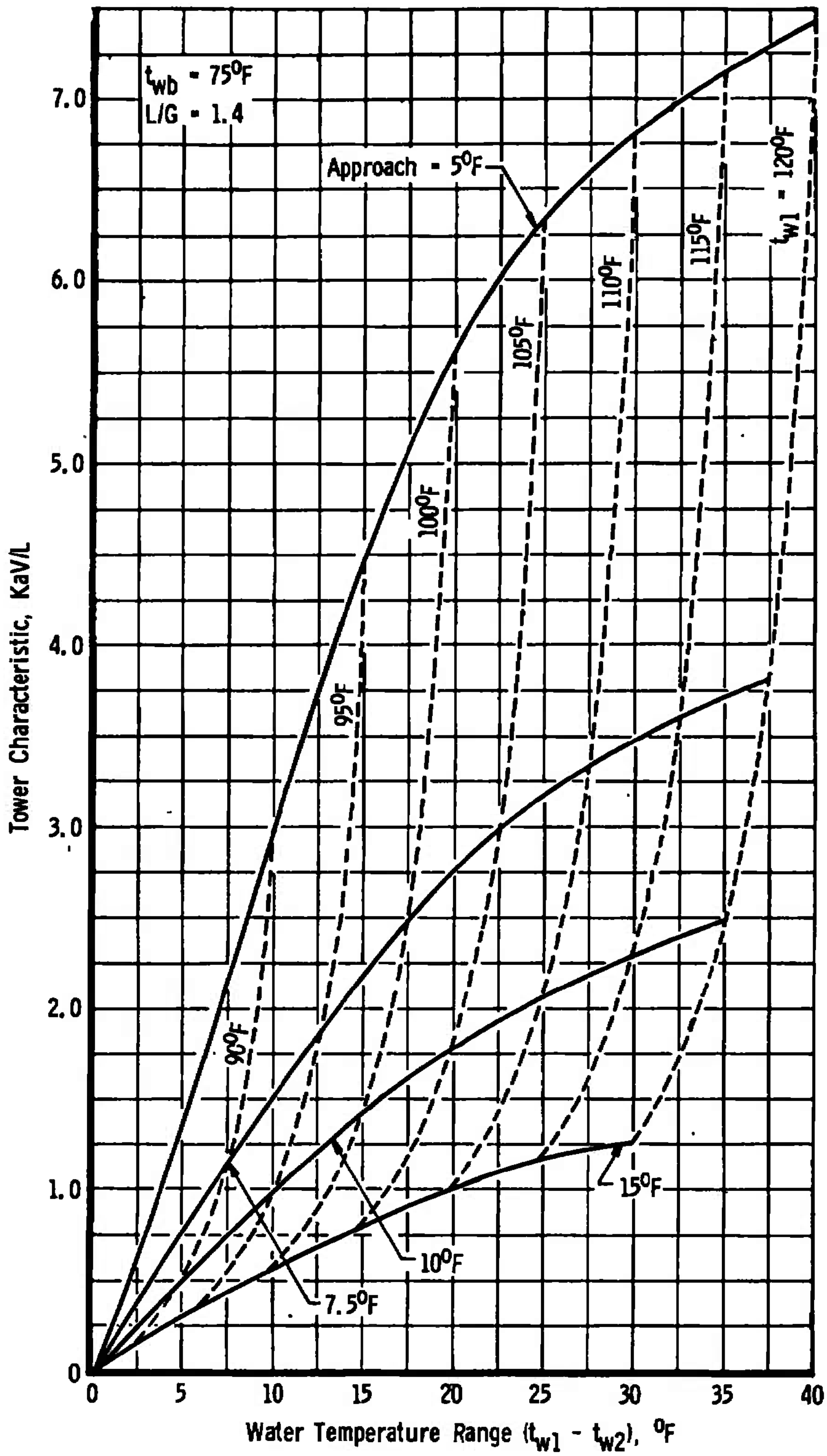


Fig. 9 Tower Characteristic versus Water Temperature Range for t_{wb} = 75°F and L/G = 1.4

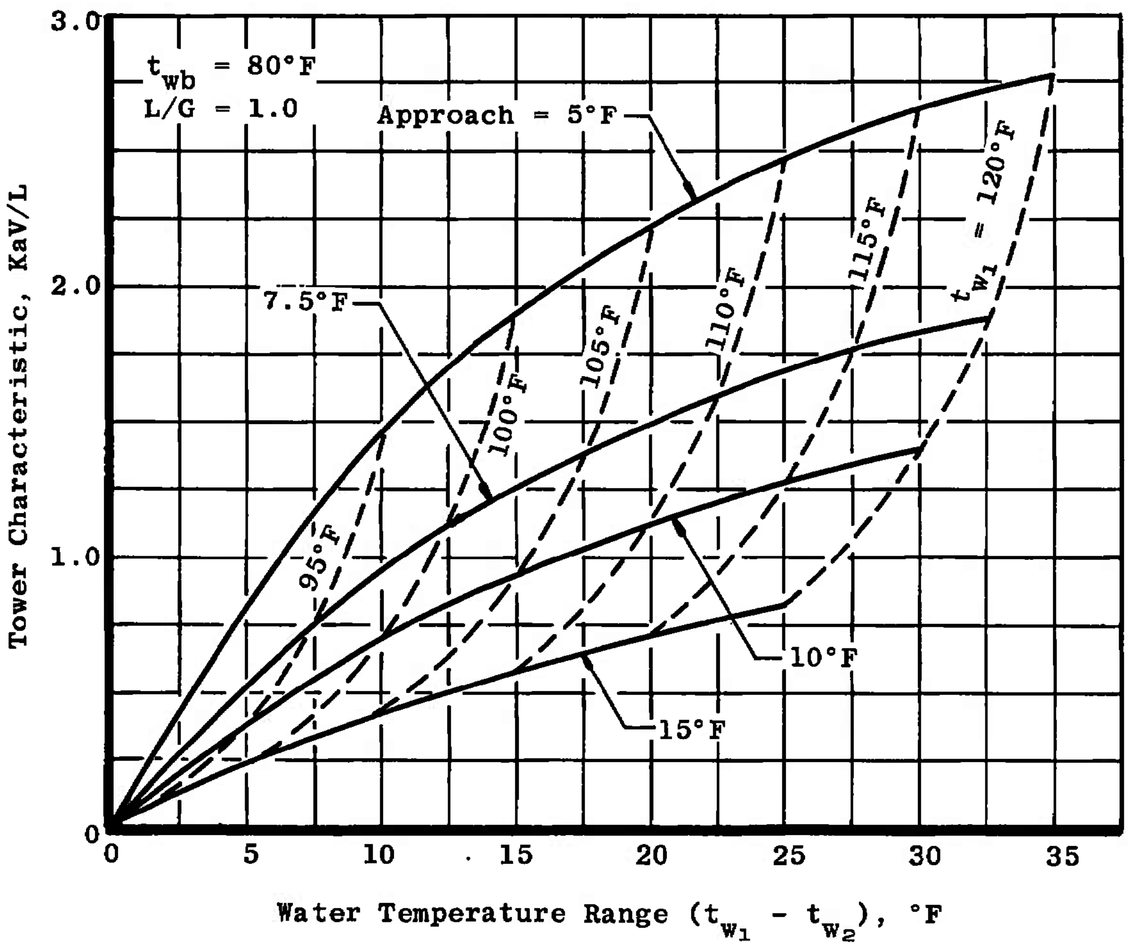


Fig. 10 Tower Characteristic versus Water Temperature Range for t_{wb} = 80°F and L/G = 1.0

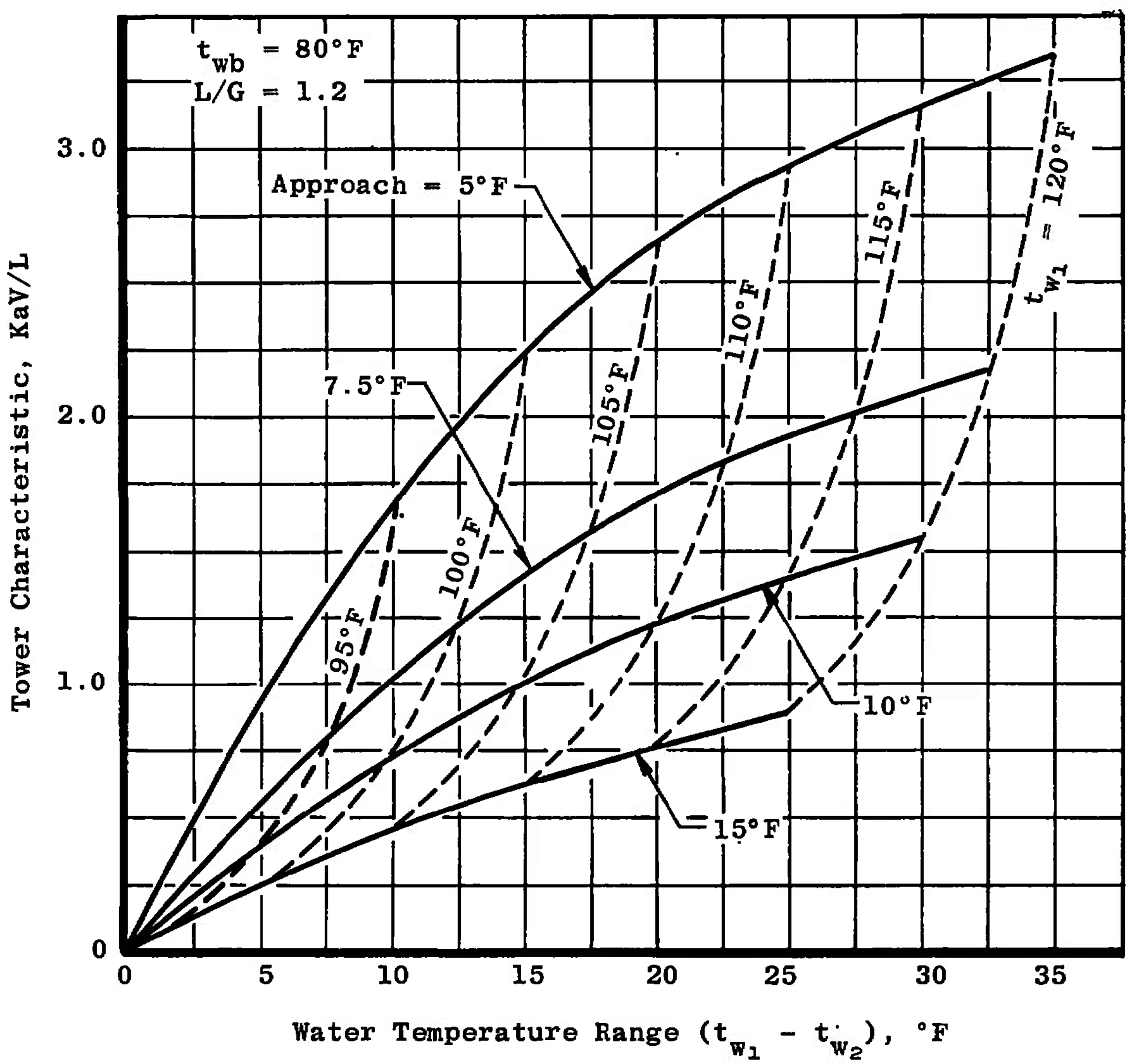


Fig. 11 Tower Characteristic versus Water Temperature Range for t_{wb} = 80°F and L/G = 1.2

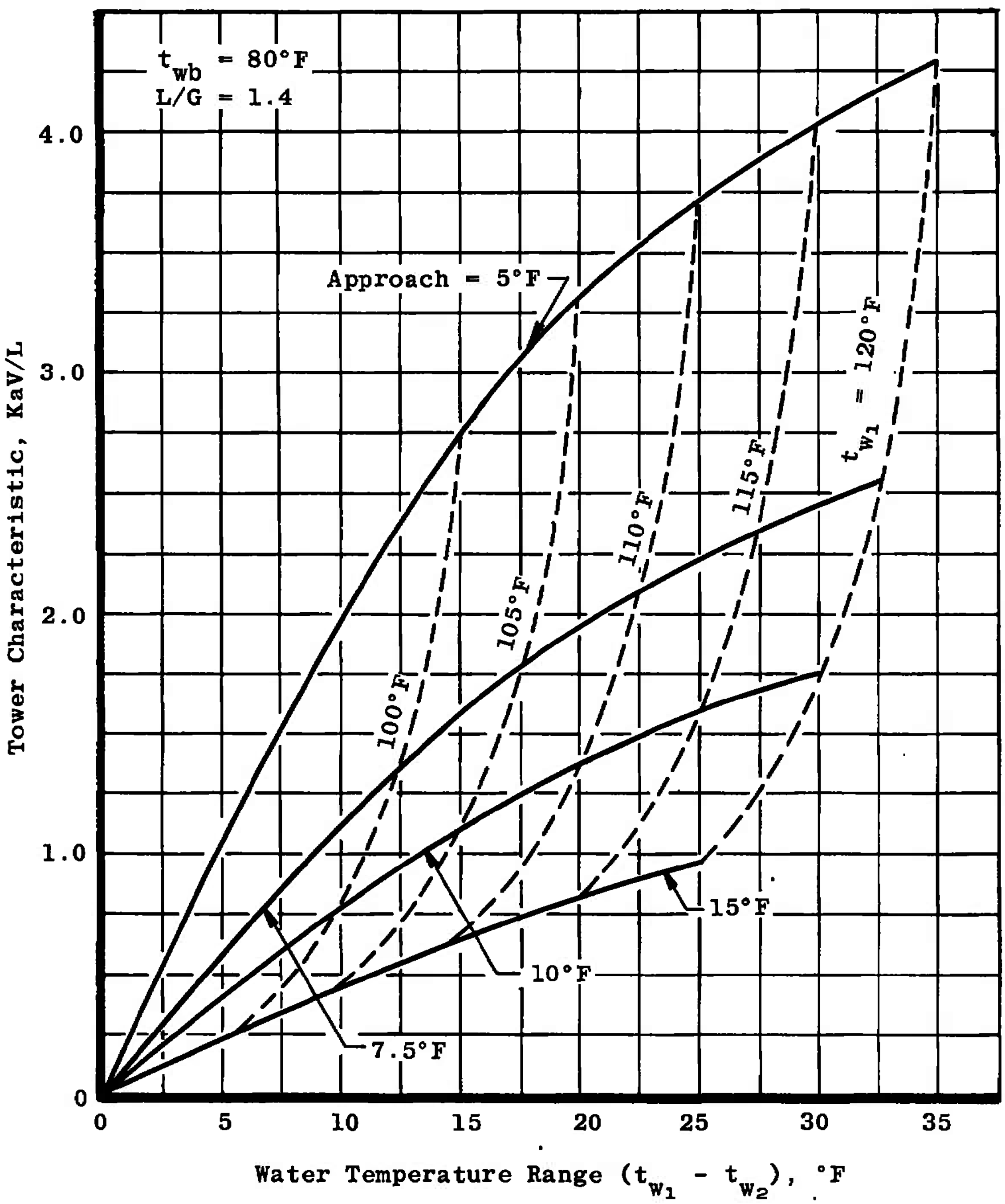


Fig. 12 Tower Characteristic versus Water Temperature Range for t_{wb} = 80°F and L/G = 1.4

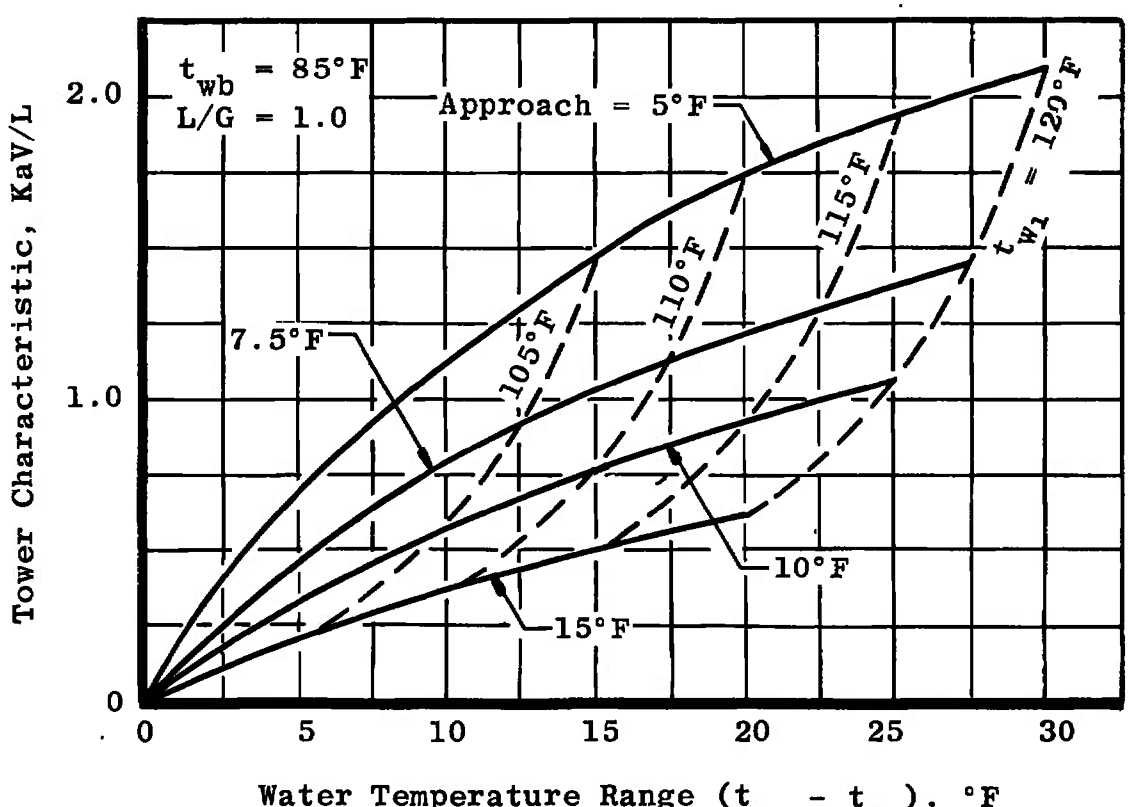


Fig. 13 Tower Characteristic versus Water Temperature Range for t_{wb} = 85°F and L/G = 1.0

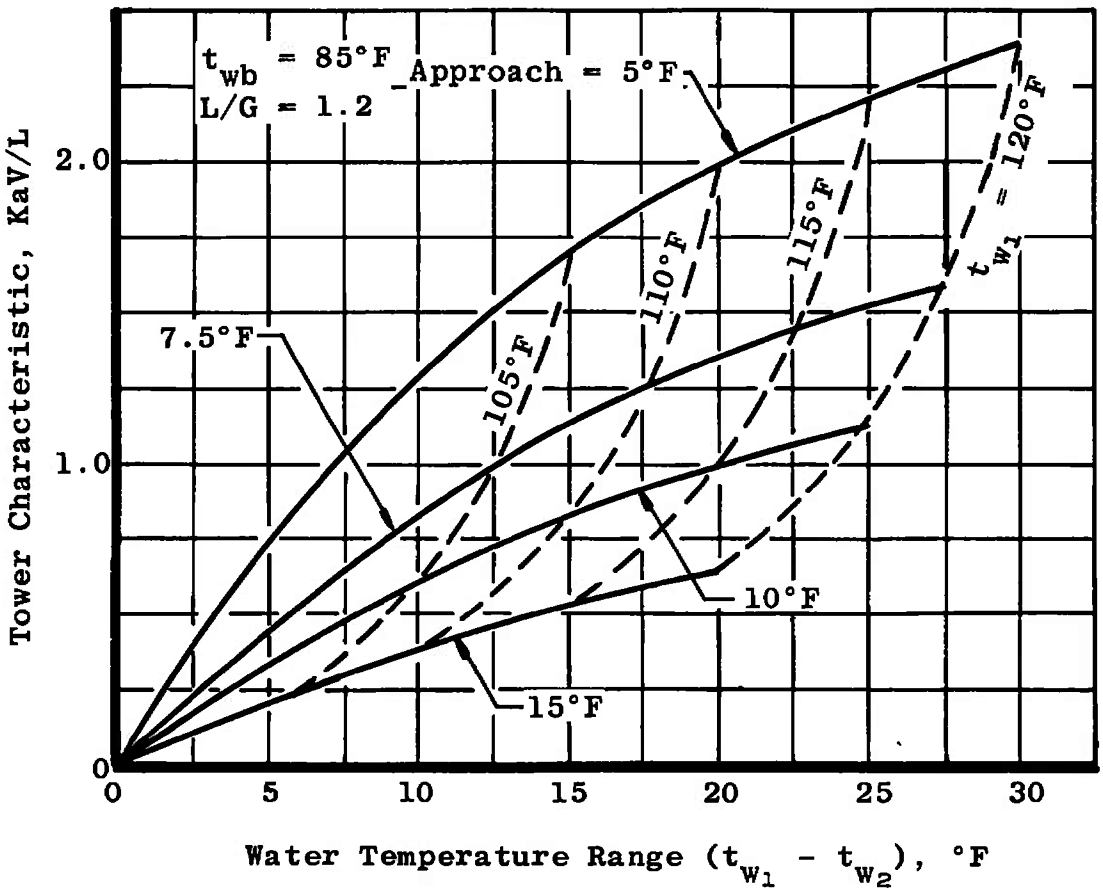


Fig. 14 Tower Characteristic versus Water Temperature Range for t_{wb} = 85°F and L/G = 1.2

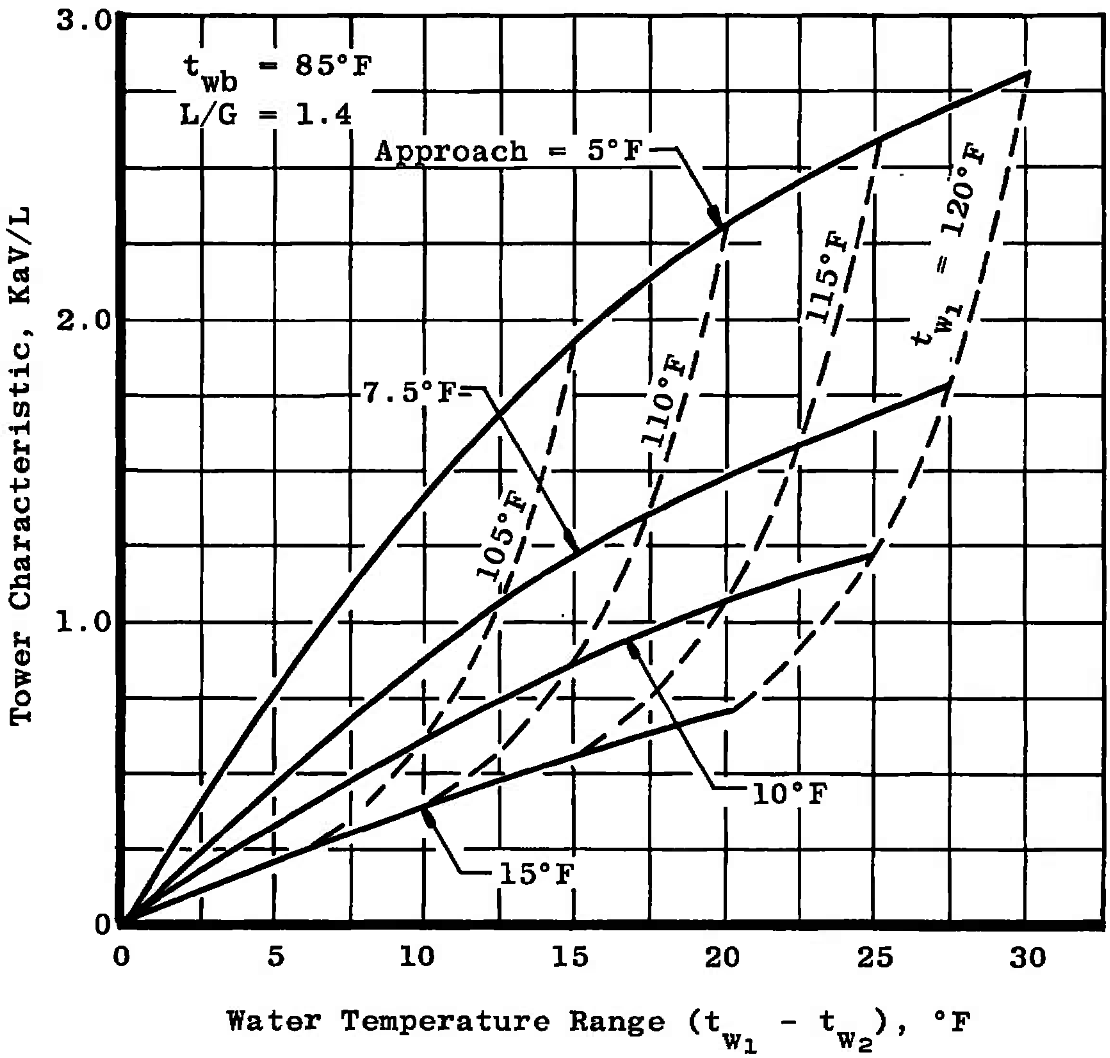


Fig. 15 Tower Characteristic versus Water Temperature Range for t_{wb} = 85°F and L/G = 1.4

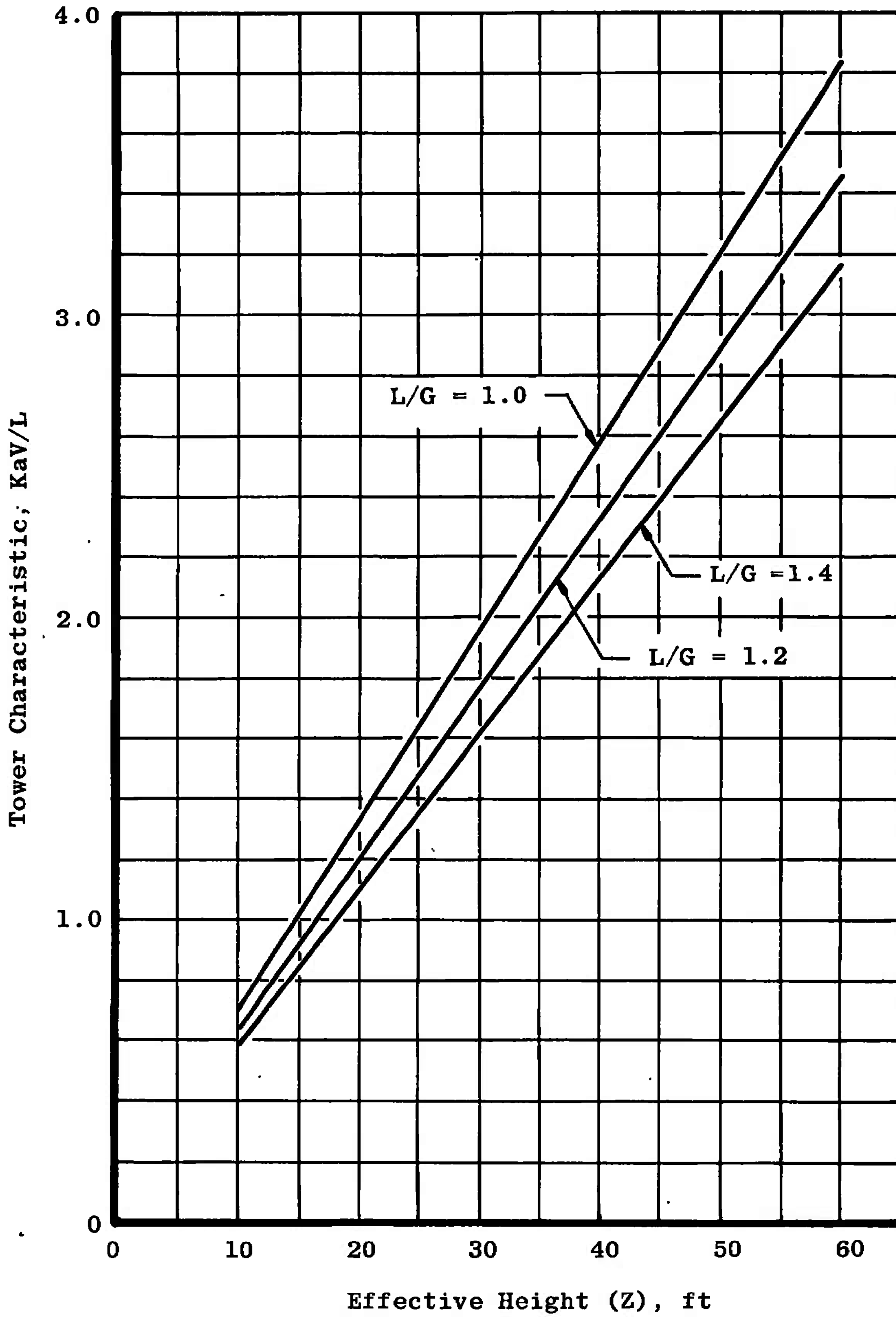


Fig. 16 Tower Characteristic versus Effective Height

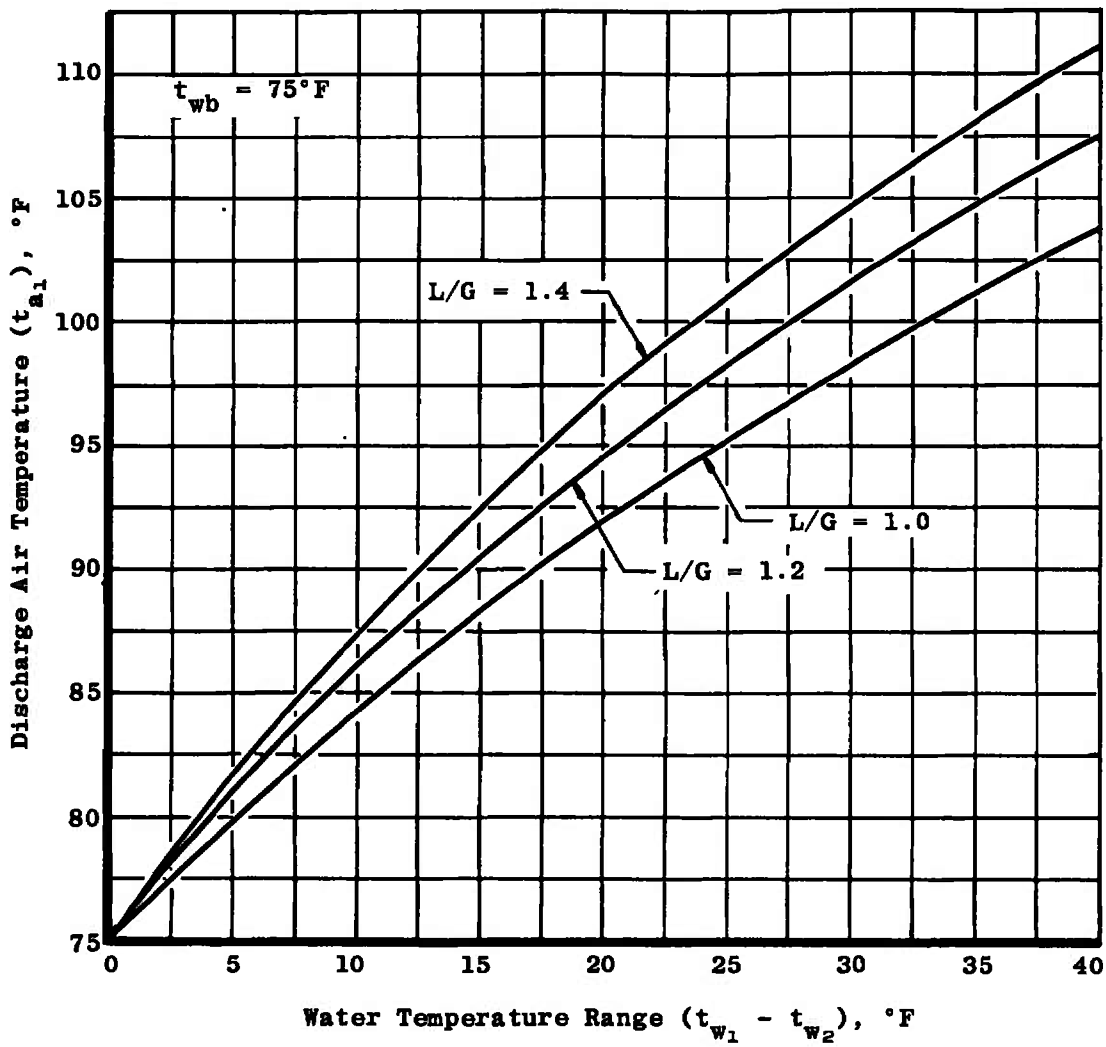


Fig. 17 Discharge Air Temperature versus Water Temperature Range for t_{wb} = 75°F

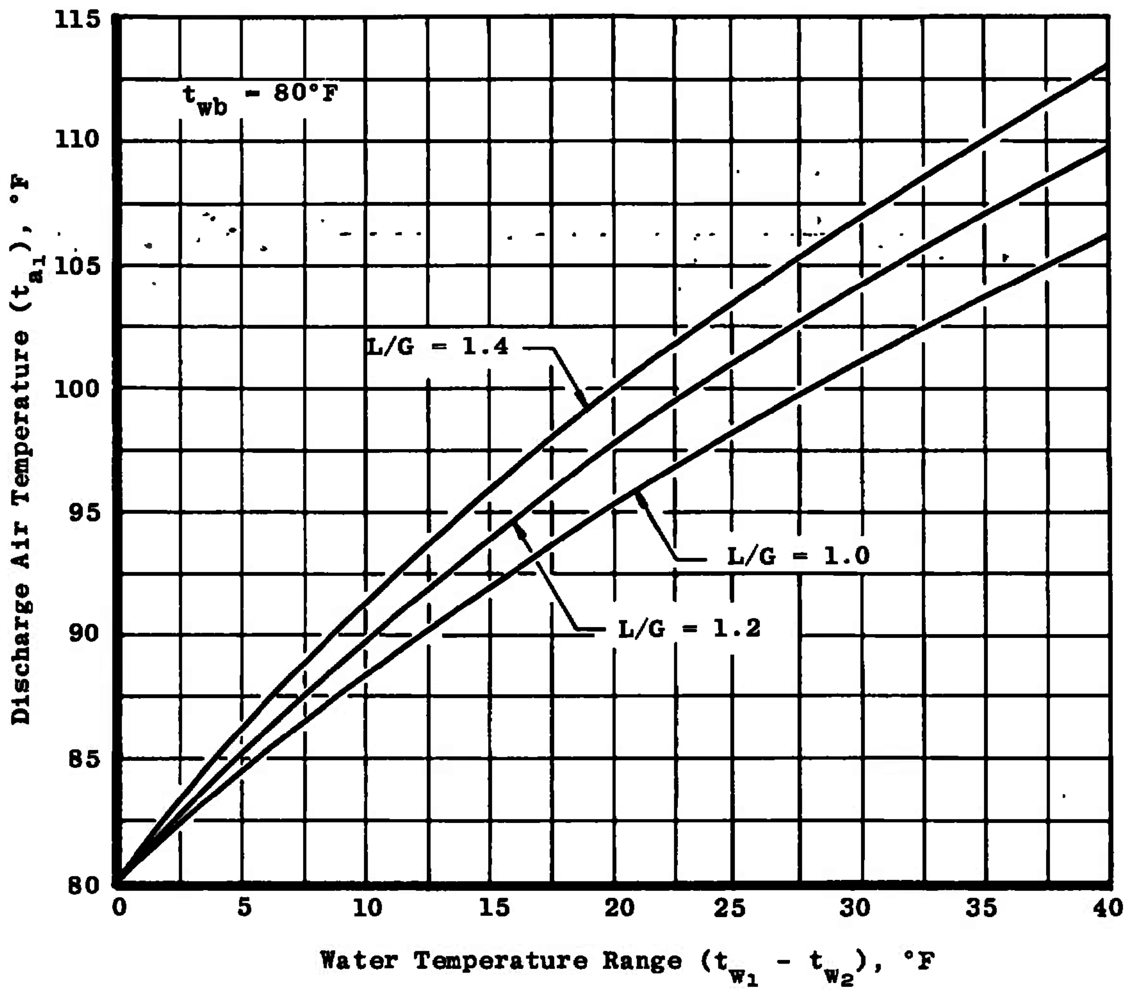


Fig. 18 Discharge Air Temperature versus Water Temperature Range for t_{wb} = 80°F

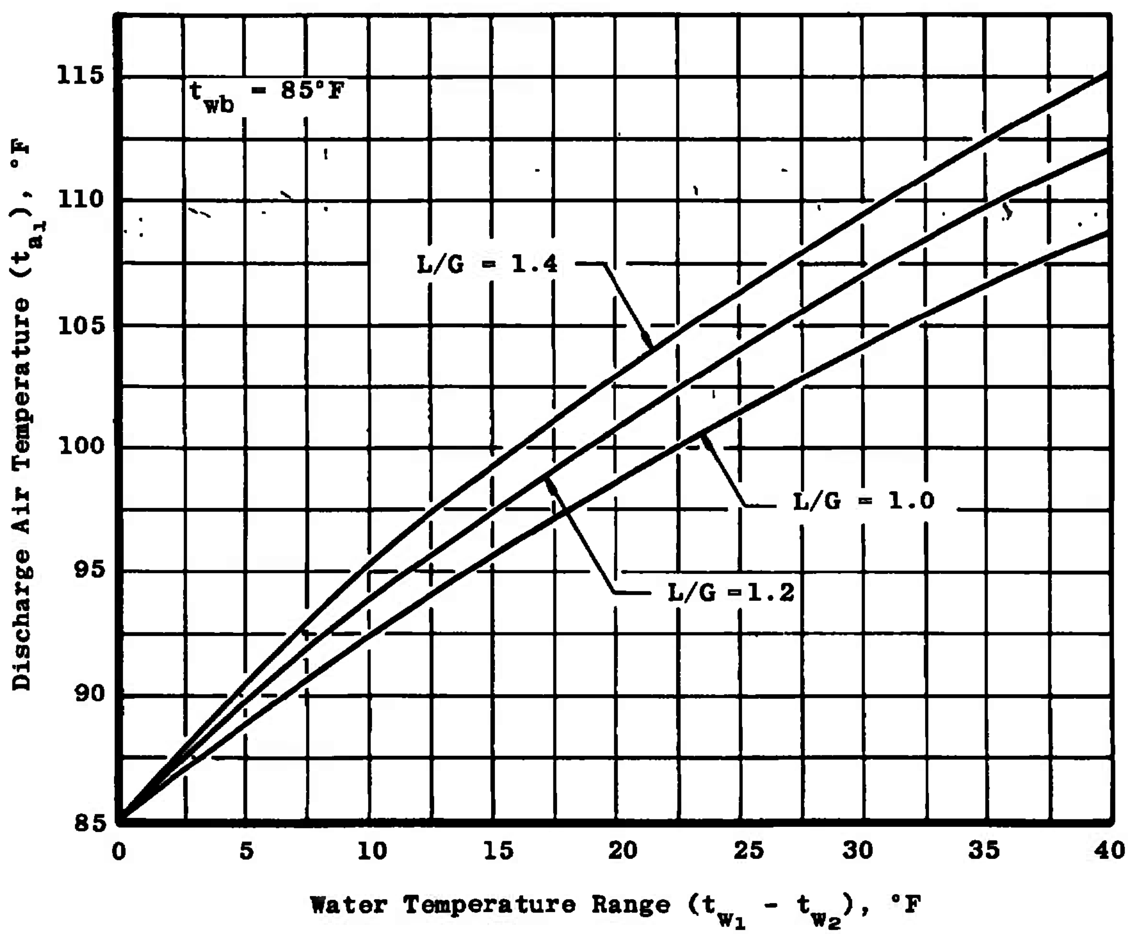


Fig. 19 Discharge Air Temperature versus Water Temperature Range for t_{wb} = 85°F

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13. ABSTRACT

The need for a method whereby the preliminary design of a cooling tower can more easily be specified has existed for many years. The numerical techniques commonly used to solve the Merkel cooling tower equation are not adaptable to a direct general solution. This report presents a direct solution to the Merkel equation which, when combined with certain detailed experimental data by others, yields a relatively quick and straightforward method of estabilishing the preliminary design of a mechanical-draft counterflow cooling tower. Special attention is devoted to cooling tower application relative to high altitude engine test facilities.

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